PATENT

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Linda Massey

Linda Massey

Applicants: Bitterly, Jack G. et al.
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Commissioner for Patents
Washington, D.C. 20231

AMENDMENT AFTER ALLOWANCE AND PAYMENT OF ISSUE FEE

Sir:

AMENDMENTS TO THE SPECIFICATION

Please replace the paragraphs of the specification beginning on the pages and lines indicated with the following replacement paragraphs:

1. **PAGE 1, LINES 16:**

Distillation is slow at atmospheric pressures unless the heat flux raises the temperature to the boiling point of water, 212°F (100°C) at sea level. (Metric conversions are approximations.) Therefore, to distill water at atmospheric pressures, heat energy must raise the temperature from ambient to 212°F (100°C). At that temperature, water boils and vaporizes, changing from liquid to gas. Once the water vaporizes, a cold source must be present to condense liquid water from the water vapor. One must use additional energy to remove heat from a cold trap and create a continuous cold source to condense the fluid.

2. **PAGE 2, LINE 3:**

Not only is higher temperature distillation expensive, it can cause an additional problem. When processing contaminated liquids that contain minerals or organic molecules, higher temperature can cause chemical reactions between the

molecules. Some reactions can form high molecular weight molecules that can obstruct boiler walls and make cleaning difficult. High temperatures also break down the walls of organic cells that makeup the contaminates themselves, which in turn can release toxic materials (pyrogens) into the liquid. High temperature boiling can cause some lower molecular-weight contaminates to vaporize and migrate with the water vapor toward the condenser.

3. **PAGE 2, LINE 13:**

Despite problems with ambient-pressure distillation and R.O., desalination capacity in the United States has increased. According to the Office of Technology Assessment, in 1955, for example, the United States had almost no capacity, and less than 30 million gallons per day (Mgal/d) (113.5 million liters per day) could be produced in 1970. By 1985, capacity exceeded 200 Mgal/d (757 million liters per day). Still, that amount is quite small compared to the annual water use in the United States. For example, the United States Geological Service reports that the overall fresh water withdrawals in the United States in 1995 were 341,000 Mgal/d (1.29×10^{12} liters per day).

4. **PAGE 2, LINE 21:**

Conventional distillation systems use conventional boilers. Boilers are an advanced art whose efficiencies have been studied and documented. *See* e.g., McAdams, W. H., *Heat Transmission* 2d Ed., McGraw-Hill 1942, pp. 133-137.

5. **PAGE 3, LINE 25:**

In units of gallons per day, the mass flow rate, $\dot{w}_G$ equals:

6. **PAGE 4, LINE 15:**

Thin boiler wall thickness: It is important to utilize a very thin boiler/condenser wall surface thickness t_{wall}, with metals that have high heat conductivity k_{wall}. Typically, the wall thickness is between 0.010 inches to 0.015 inches (0.25 mm -- 0.38 mm). The heat conductivity for steel, a typical boiler wall surface, is about 25

BTU/(ft hr °F) (0.43 Watt/cm°C), which yields a boiler wall conductivity heat transfer rate of between 20,000 and 30,000 BTU/(ft^2 hr °F).

7. PAGE 4, LINE 21:

Thin Film Boiling: Minimizing the contaminated water fluid film thickness against the boiler surface improves heat transfer to the fluid in a boiler. Conventional boilers do not create a uniform thin fluid film against the boiler surface. Consequently, they must rely on high temperature gradients to conduct heat through the fluid film. Thin film boiling normally operates at lower boiling temperature. In the prior art, the thin film of liquid is deposited along the boiler wall in two different ways, wiping or spraying.

8. PAGE 5, LINE 11:

High boiler heat transfer: Under conditions of relatively low fluid velocities (low Reynolds numbers, Re), convection and phase change processes govern the boiling heat transfer rate. Thin film conditions in the boiler create a condition whereby nucleate boiling can occur at low ΔT, which produces high heat transfer rates. The heat transfer process with phase change is more complicated than the normal liquid-to-liquid phase-only convection process. In liquid-phase convection, one can describe the methodology by including the fluid effects of viscosity, density, thermal conductivity, expansion coefficient, and specific heat along with the geometry of the system. However, the mathematics for heat transfer with a phase change also includes the surface roughness characteristics, the surface tension, the latent heat of evaporation, the pressure, the density and other properties of the liquid-vapor. The entire process becomes so complex that empirical experimental data and dimensional analysis determines the analytical expressions.

9. PAGE 6, LINE 9:

The principal object of the present invention is to disclose and provide an efficient water reclamation system that can distill brine and other contaminated liquids at low cost. By reducing the system's operating pressure to near the vapor pressure of the contaminated fluid, boiling can occur at ambient temperatures with a small

temperature difference (ΔT), e.g., 6°F (3°C) or less. By having low temperature boiling occur on one side of a thin boiler wall and condensation occur on the other side of the wall, heat transferred to the wall by the condensing fluid can provide energy for boiling on the opposite side of the wall. Further, constructing the boiler/condenser wall as a cylindrical shell, having the boiling surface face toward the axis of rotation of the cylinder and then rotating the cylinder about the axis, heat transfer for boiling improves due to the higher g forces on the fluid. Likewise, because the condenser surface faces outward, as vapor condenses, the drops of condensate are thrown off the condenser surface. That leaves a clean surface to receive vapor, which improves condenser efficiency. Because of the high heat transfer capabilities possible from a rotating boiler/condenser, boiling can occur practically at ambient temperatures Low temperature boiling minimizes scale and fouling of the boiling wall surface. This also allows the boiling heat transfer rate to be maintained at a very high level.

10. PAGE 7, LINE 5:

Some or much of the contaminated fluid likely will not vaporize. Thus, the system may not convert 100% of incoming brine, for example, to potable water. As the percentage converted increases, the boiling point rise increases, and the energy requirements of the system increase. This occurs because as the salinity concentration increases a greater temperature increase is required to boil the remaining fluid. Brine from seawater is inexpensive, and returning brine back to the ocean at slightly higher salt concentrations usually is acceptable. Accordingly, an object of the present invention is to allow it to keep the percentage of brine converted well below 100%, use less energy and be very economical. On the other hand, where the system is removing toxic wastes that must be stored or disposed, limiting the volume of the output (i.e., limiting the amount of fluid that remains with the contaminate) probably is desirable. Therefore, these systems may process and vaporize a higher percentage of these contaminated fluids. Accordingly, another object of the

present invention is to design a system that can process different types of fluid from brine to highly toxic waste.

11. **PAGE 8, LINE 6:**

One end of each boiling space is open to a compressor, which raises the pressure and adds heat to the water vapor after the water boils. As fluid boils, the compressor transfers the vapor to the condensing spaces, which are open to the downstream side of the compressor. The vapor strikes the condenser wall, which is a wall that is common to the adjacent boiling space. The vapor condenses and transfers its heat to the wall or shell. That heat energy boils new, incoming liquid. This minimizes the heat energy that the system uses. In other words, the latent heat of condensation in the condenser is transferred and then used as the latent heat of vaporization in the boiler. The system needs no heat sources other than the energy of vapor compression to complete the cycle flow from vaporization to condensation.

12. **PAGE 8, LINE 25:**

As the liquid boils in the boiler, the high g forces maintain the contaminates on the boiler wall. The system's construction also prevents the contaminates from flowing back toward the inlet. High g forces also cause the contaminates to flow along the boiler wall. When they reach the end of the wall, they are thrown outward and collected in a ring. From the ring, the contaminates are pumped or otherwise directed out of the housing. To keep the sludge and pure liquid separate, collection of contaminates and pure liquid can occur at opposite ends of the shells.

13. **PAGE 9, LINE 18:**

FIG. 5 is a perspective view of the rotating boiler/condenser showing the condenser inlet side of an exemplary embodiment of the water reclamation system of the present invention.

14. **PAGE 9, LINE 26:**

FIG. 9 is a detailed sectional view of the processed waste end of an exemplary embodiment of the present invention.

15. PAGE 10, LINE 4:

FIG. 13 is a perspective view of an array of 30 such vertically mounted systems.

16. PAGE 10, LINE 7:

FIG. 15 is a perspective view of part of a shell for another embodiment of the present invention.

17. PAGE 10, LINE 23:

FIG. 27 is a graph showing how the cumulative boiler area varies as a function of the radius of the outer shell assuming a given shell separation, inner shell radius, shell thickness and shell length.

18. PAGE 10, LINE 28:

FIG. 29 is a series of graphs showing energy usage per output pound of water for different boiler temperatures and variable boiler-to-condenser temperature differences ΔT.

19. PAGE 11, LINE 1:

FIG. 30 is a series of graphs showing energy usage per output pound of water for differing boiler-to-condenser ΔT temperature difference values based on different combined motor compressor efficiencies.

20. PAGE 11, LINE 9:

The water reclamation system **10** of the present invention uses low-pressure boiling and condensation, which FIG. 14 shows in simplified form. The system is conventional. It has an evaporator or boiler chamber **12** and a condenser chamber **14**. Both mount in a vacuum chamber (not shown in FIG. 14) so that the contents in the boiler and the condenser are under vacuum. Contaminated liquid or brine **16** is fed into the boiler chamber where it pools on common boiler/condenser wall **18**. This application uses the terms "contaminated liquid" or "brine" interchangeably.

21. PAGE 12, LINE 6:

The reclamation system **30** of the present invention comprises an inner housing **50** that mounts within a vacuum chamber **80** (FIG. 1). The diameter of the inner housing **50** of the exemplary embodiment is about 14.1 in. (36 cm). The entire system may be scaled up or down, however. As is explained in more detail, the system **30** comprises a series of concentric metal shells. In the exemplary embodiment, the shells are stainless steel, aluminum or other metal although some plastics may be acceptable. Metals have better heat transfer capabilities, but plastics are less expensive to fabricate.

22. PAGE 12, LINE 13:

The exemplary embodiment of FIG. 1 has four shell sub-assemblies, and each sub-assembly comprises two shells. Only three of the shells (of a typical boiler-condenser sub-assembly) are discussed initially—outer shell **40**, intermediate shell **32**, which is within the outer shell **40**, and inner shell **34**, which is within the intermediate shell, FIGS. 1-4 and 10. "Inner," "intermediate" and "outer" are relative terms and represent three adjacent shells in the inner housing **50**.

23. PAGE 12, LINE 18:

The space between the outer shell **40** and the intermediate shell **32** is a boiling or vapor chamber **38**. Similarly, the space between the intermediate shell **32** and the inner shell **34** is a condenser chamber **36**. In this nomenclature of FIG. 1, the adjacent boiler-condenser chambers are neighboring two different boiler-condenser sub-assemblies above and below. In this embodiment, the radial height of each chamber is 0.395 in. (10 mm).

24. PAGE 12, LINE 22:

The shells mount within the inner housing **50** (FIGS. 1-4). Though the inner housing can be metal, it is a rigid plastic such as Lexan® in the exemplary embodiment. The inner housing is formed of two housing halves **52** and **54** that attach together along annuluses **56** and **58**. The assembly also can be split at locations **280** and **300**, each location having appropriate fastening adhesive, welding, bolts or

other fasteners to secure the annuluses together. The connection should be secure and air tight because the inside of the inner housing **50** is at a different pressure than outside the housing **50**. The diameter of the inner housing in this exemplary embodiment is approximately 14.2 in. (36 cm), and its length is about 11.5 in. (29.2 cm). These dimensions will change if the system is scaled up or down. The inner housing halves are preferably identical to decrease the fabrication costs.

25. PAGE 13, LINE 3:

The shells **32**, **34** and **40** are formed of high heat conductivity metal such as stainless steel. Stainless steel also is strong, relatively low cost and is not corroded by salts or contaminants. Anodized aluminum also is a possible choice for the material. It weighs less than stainless steel and has better heat conductivity. It is weaker, however, and more corrosive. Though other metals are possible, some are too weak, more expensive, not good conductors of heat or not corrosion resistant. The size of the unit may be a factor in material choice.

26. PAGE 13, LINE 10:

This application gives special attention to the inner and outer surfaces of the shells (e.g., shell **34 or 36**) to provide near ideal boiling and condensing. Boiling surfaces in particular may be treated with special surface coatings and these have been examined for their applicability to this invention. For reasons that are discussed below, the shell material may have grooves scratched into the surface. A non-wettable coating of a plastic material such as Teflon® may enhance boiling or condensation.

27. PAGE 13, LINE 16:

The inner housing **50** mounts for rotation within the outer housing or vacuum chamber **80**. A substantial pressure force (maximum of 1 atmosphere) exists between the internal vacuum in chamber **80** and the external ambient pressure. Therefore, the walls **88** of the vacuum chamber must be thick enough to resist crushing from those pressure forces outside the chamber. The wall thickness is calculated using known relationships with adequate safety factors and must be thicker

for larger-diameter chambers. To save weight and material, the outer wall **88** may be "pocketed" similar to an egg container. Other strengthening techniques also may be used.

28. PAGE 13, LINE 24:

A pair of end caps **84** and **86** and a center cylindrical section **88** form the outer housing and the vacuum chamber **80** (FIGS. 1-4 and 11). The outside diameter is approximately 19.7 in (50 cm)Again, the end caps and the cylindrical section may be metal, but they are a rigid plastic such as Lexan® in the exemplary embodiment. Plastic is strong enough to resist the load on the small diameter system that FIG. 1 shows. Larger vacuum chambers may require metal construction or be reinforced by wound fiber composite. Each end cap has a frusto-conical portion **90** that curves into a narrow cylindrical portion **92** (FIGS. 1 and 11). The flat end **92** allows parts to be attached to each end cap as FIG. 1 shows. Those parts are discussed below. The end-to-end distance between the outside of each end cap is about 25.6 in (65.0 cm).

29. PAGE 14, LINE 3:

The system may use axial fences fastened to the boiler or condenser surfaces to control the filming or otherwise control the fluid. See FIGS. 15 and 16 for an alternative shell construction that has a form of a fence.

30. PAGE 14, LINE 10:

The cylindrical section **88** of the vacuum chamber **80** could be molded plastic, wound fiber composite or metal. Strengthening ribs **100** face outward from the cylindrical section (FIG. 11). Annular rings **102** on the end caps **84** and **86** connect to the annular rings **104** on the cylindrical section **88**. Bolts **107** (FIG. 2) through the annular rings secure the cylindrical section to the end caps, but other fastening is possible. The exemplary embodiment uses bolts because they can be removed for access into the chamber. The end caps **84** and **86** also have an annular tongue **104** that seats up against the annular grooves **106** on the cylindrical section. The tongue and groove align the end caps to the cylindrical section and help create a better

seal. An O-ring (FIGS. 1-3) between the tongue and groove assures the vacuum seal. Because the inside of chamber **80** is at near-vacuum pressure, ambient pressure tends to push the end caps against the cylindrical section.

31. PAGE 14, LINE 26:

The inner housing **50** mounts for rotation within the chamber **80** about an axis of rotation **132**. In the exemplary embodiment, hollow shafts **120** and **122** extend through the respective hubs **96** and **97** of end caps **84 and 86** along the axis of rotation. The shafts are hollow because they carry fluid as discussed below. Each shaft has a flange **124** and **126** that is secured in respective recess **128** and **130** (FIGS. 1 and 10). Because the shafts support the inner housing **50** for high speed rotation, they must be strong, precision parts, and maintaining tolerances for mounting them in the hubs **96** and **97** of chamber **80** also is important (FIG. 10).

32. PAGE 15, LINE 3:

Shaft **120** on the left side (FIGS. 1 and 9) of the system extends into hub **140** on the end **142** of inner housing half **52** as best shown in FIG. 9. A bearing **146** permits rotation of the hub **140** (and the inner housing **50**) about the axis of rotation **132**. Similarly, as best shown in FIG. 10, shaft **122** on the right side of the system extends into inner housing hub **148**. A bearing **152** permits rotation of the hub **148** about the axis of rotation. Bearings **146** (FIG. 9) and **152** (FIG. 10) are ball bearings in the exemplary embodiment, but those skilled in the art may substitute other bearings such as magnetic bearings. Any bearings should be long-lived and have no corrosion. The size and weight of the system and the rotational velocity of the inner housing **50** will influence the choice of bearings.

33. PAGE 15, LINE 12:

A motor **160** rotates the inner housing **50** by belt, chain or gear drive (FIG. 1) in the exemplary embodiment. The motor mounts on the outside of the right-side end cap **90**. As FIG. 10 also shows, the motor extends inside the outer housing **80**. Both end caps **84** and **86** have structure **168** (FIGS. 10 and 43) for mounting the

motor so that the end caps can be identical to save on fabrication costs. However, the system uses only one motor **160**, and it mounts on the right-side end cap.

34. PAGES 15, LINE 25:

In the belt driven system of the exemplary embodiment, the motor **160** drives a pulley **162**, which drives a belt **166**. The belt, in turn, drives a pulley **164** attached to the hub **122** on the right side of the inner housing **50**. FIGS. 1, 4 and 10. Applicants use a belt drive for its simplicity and because it minimizes vibrations. A direct, chain, gear or other drive also could be used. The output speed of the motor and the relative diameters of pulleys **162** and **164** affect the rotational velocity of the inner housing **50**. Applicants anticipate that for a small system such as that shown in FIG.1 and other related figures (e.g., about 350 gal./day), the inner housing should rotate at about 1,000 rpm, which generates g-forces on the outer-most shell of roughly 50 g's.

35. PAGE 16, LINE 3:

Fluid and Mass Flow: Contaminated fluid enters the system from the right side (FIG. 1) through inlet tube **170** (FIGS. 1 and 10). The inlet tube is within shaft **122**. The upstream end **172** of inlet tube **170** is stationary and connects to a source of brine or other contaminated liquid. The downstream end **174** of the tube, however, rotates with the rotation of the inner housing **50**. Therefore, a seal (not shown) is necessary between the upstream and downstream ends of the inlet tube **170**. The seal can be outside the vacuum chamber **80**, inside the hub **97** of the chamber or inside the chamber.

36. PAGE 16, LINE 10:

The downstream end **174** of the inlet tube terminates into several branches. The exemplary embodiment has three such branches **180**, **182** and **184**. See FIG. 5 in particular. The number of branches may vary depending on the angular velocity, the size of the shell housing and the number of shells. Each branch has several injectors—four **186**, **188**, **190** and **192** (FIG. 10) in the exemplary embodiment for each branch. See FIG. 5. The exemplary embodiment uses four injectors per branch

because each injector aligns with one of the boiling or vapor chambers such as chamber **38** (FIG. 1). The exemplary embodiment has four such chambers.

37. PAGE 16, LINE 18:

Contaminated liquid flows from the inlet tube **170**, through a seal, into the branches **180**, **182** and **184** and then through the injectors **186**, **188**, **190** and **192** and into the boiling chambers, e.g., chamber **38**. The high velocity rotation of the branches creates a head of pressure on the contaminated liquid that varies with the injectors' distance from the axis of rotation **132**. Consequently, each injector may have a flow restrictor or nozzle to compensate for the pressure differences. On the other hand, when the contaminated liquid is injected into the boiling chamber, the centrifugal force from rotation creates a thin film of contaminated liquid along the inside of the shell **40**. Because the area of the inward-facing wall farther from the axis of rotation is greater than the area of a similar wall closer to the axis, the farther wall requires more contaminated liquid to yield the same thin film thickness. Fluid viscosity also affects flow rates. The nozzles or other flow controls can be adjustable under computer feedback control to account for changing conditions.

38. PAGE 16, LINE 30:

Because the contaminated liquid is at low pressure, heat energy transferred across the shell wall causes the liquid to boils at the low temperature. As discussed previously, the thin film and high centrifugal forces enhance boiling. Some of the contaminated liquid becomes vapor. This vapor is pure, uncontaminated gaseous molecules. In conventional boilers, some contaminates vaporize or are mechanically thrown off as solids and become part of the distillate. The g forces acting on the contaminates in the present invention prevent the denser molecules from leaving the film of contaminated liquid. Consequently, the condensed water is purer than one could obtain with conventional boilers and condensers and likely will not contain bacteria, viruses, organic molecules or metals. The system should develop sufficient g forces , even some viruses may not vaporize.

39. PAGE 17, LINE 12:

Before following the sludge, the flow of vapor is discussed. Annular walls **200**, **202**, **204** and **206** (FIG. 5) close the upstream end of the boiling chambers such as chamber **36** (FIG. 2). The injectors **186**, **188**, **190** and **192** (FIG. 5) pass through the annular walls, but the walls block the flow of contaminated liquid, sludge or vapor back to the upstream side of the boiling chambers. Thus, the contaminated liquid, sludge and vapor move to the left in FIG. 1.

40. PAGE 17, LINE 18:

After the vapor exits the downstream side of the boiling chambers (left side in FIG. 1), e.g., chamber **38**, a compressor fan **220** draws the vapor. Chamber **38** is at or below 0.1 atmospheres, or close to near vacuum because of the very low pressure in outer vacuum chamber **80**. Fan **220** also lowers the pressure within the condensing chamber **38**.

41. PAGE 18, LINE 1:

The vapor exits the fan **220** and enters a duct **226** behind the fan (FIGS. 1-4). The shape of the duct will yield a desired pressure at the fan exit to achieve chosen pressure ratios. In the exemplary embodiment, the diameter of the inlet to the duct at the fan is 5.6 in (14.2 cm) and the diameter at the outlet is 4.8 in (12.2 cm), a 1:0.86 ratio. The operating conditions including the ambient temperature and actual contaminated liquid affect these ratios. Any adjustments should act within the fan's efficiency goals.

42. PAGE 18, LINE 7:

Returning to the description of the boiler and condenser process, the fan slightly compresses the vapor from the boiling surfaces, which causes adiabatic heating of the vapor. The amount of heating should be less than 6 °F (3 °C). The slightly heated vapor then flows out the right side of duct **226** (FIG. 1) and into the condensing chambers such as chamber **36**. There, the vapor encounters the outside surface of shell **34**. When the contaminated liquid boils on the inner surface of shell **34**, heat energy from the shell transfers to the contaminated liquid. Accord-

ingly, the shell wall cools. When the slightly pressurized and warmer vapor (from fan **220**) strikes the cooler shell wall, it condenses.

43. **PAGE 18, LINE 20:**

This condensate pools along the outer shells of the condensing chambers, such as shell **32** of chamber **36**. Centrifugal force urges the pure water into a thin film. Blocked from moving to the right by walls such as wall **200**, the pure water flows toward the left ends (FIG. 1) of the shells. Thus, in the FIG. 1 embodiment, the pure water collects on the left side of the condensing chambers such as chamber **36**. The sludge also moves to the left side but of the boiling shell such as shell **38**.

44. **PAGE 18, LINE 26:**

There are many ways to collect the condensate, and this application discusses several. Turning first to the exemplary embodiment of FIG. 1 and as seen in FIGS. 1-4 and 6, as condensate flows to the left (FIG. 1) it reaches a wall **240**, **242**, **244** or **245** (FIG. 6). Each of these condensing chamber walls has an outlet tube **246**, **248**, **250**, and **254** (FIG. 1) extending through the wall from the condensing chamber to a collector tube **256**. The exemplary embodiment has three sets of these collector tubes **256**, **258** and **260** (FIG. 6). As FIG. 1 shows, the pure water from outlet tubes **246**, **248** and **250** reaches the collector tube. Centrifugal force from the rotating inner housing forces the pure water to tube **254** (FIG. 1). That tube passes through wall **245** (FIG. 6). Pressure from the centrifugal force in collector tubes **256**, **258** and **260** forces the pure water through tube **254** and into collector chamber **268**.

45. **PAGE 19, LINE 14:**

A stationary dipper tube **286** extends from an open end **288** in the trough **280** to a fitting **290** at the hollow shaft **122**. The dipper tube is fixed to the shaft and does not rotate with the inner housing **50**. The open end **288** faces the direction of rotation of the inner housing. Therefore, the pure condensate enters the open end at a high velocity and tends to flow in the dipper tube from the open end toward the

fitting and hollow shaft. The dipper tube is shaped to facilitate flow of condensate toward the hollow shaft **122**.

46. PAGE 19, LINE 20:

The central opening **292** of the hollow shaft **122** has a larger diameter than the outside diameter of the inlet tube **170** (FIGS. 1 and 10). That central opening extends to the fitting **290** so that the pure condensate flows through the central opening where it is collected at the end of hollow shaft **122**. A small pump may be necessary to overcome possible slight kinetic losses on the condensate from inside the system to the final collection at atmospheric pressure.

47. PAGE 19, LINE 26:

Meanwhile, the sludge, i.e., the more concentrated contaminated liquid that has not vaporized, is moving to the left (FIG. 1) along the inside facing walls of the boiling chambers (e.g., wall **40** of boiling chamber **36**). Note that the shells also taper with the larger diameter on the left side. Centrifugal force, therefore, causes the sludge to flow to the left. When the sludge reaches the left ends of the shells, the centrifugal force throws the sludge outward where it collects in a circumferential trough **300**. Centrifugal force causes the concentrated brine or sludge to pool in the trough.

48. PAGE 20, LINE 3:

A stationary dipper tube **302** for the sludge, which is similar to the other dipper tube **286** for the pure condensate, extends from an open end **304** in the dipper tube **286** to a fitting **306** at the hollow shaft **120** (FIGS. 1 and 9). The dipper tube **302** also does not rotate with the inner housing **50**. The open end **304** faces the direction of rotation of the inner housing so that the sludge enters the open end at a high velocity and flows toward the fitting **306** and hollow shaft **120** (FIG. 9). The dipper tube is shaped to facilitate flow of concentrated brine or sludge toward the hollow shaft.

49. PAGE 20, LINE 14:

The shells taper in the exemplary embodiment so that centrifugal force acting on the liquid enhances the fluid flow toward the larger diameter end. Alternatively, the shells may be cylindrical. Fluid still would flow with cylindrical walls. Because the contaminated liquid forms a thin film on the boiler walls (e.g., the inside of outer wall **40** (FIG. 1)), as more contaminated liquid is injected into one end of the shell, the g forces create a thin, even liquid level. That causes liquid to flow toward the opposite end of the shell wall, in a sense making the liquid level even. In fact, the g forces should be sufficient to cause liquid to flow upward along the wall even if the shells are mounted vertically (i.e., vertical axis of rotation **132** such as in FIGS. 12 and 13)). Vertical mounting may enhance migration of sludge downward along the boiler wall (in the direction of gravity) so it can flow off the bottom end of each shell for collection.

50. PAGE 21, LINE 7:

As the water from the incoming brine or contaminated liquid vaporizes, salts or sludge remain on the boiler surface of the shell (e.g., shell **40**). Because of the low temperature at which the system of the present invention operates, heat will not cause chemical reactions of the contaminates. Still, some sludge or salt may tend to collect on the boiler surface. Further, some incoming contaminated fluids may be quite viscous. Similarly, the process may vaporize enough water from the contaminated fluid that the resulting sludge is viscous. Accordingly, one or more high pressure spray nozzles direct water at each boiler surface of the respective shells. FIGS. 1-9 show one such spray nozzle **320**. The nozzles can be aimed permanently, or computer control can aim the nozzles, moving them to direct a pulsing spray against the wall as the wall rotates in front of the nozzles. This pulse spray loosens sludge so that centrifugal force carries the sludge to the left and off the end of the shells. In FIG. 1, the nozzle is on the left side of the rotating shells, but it can be on the right side. Alternatively, a single high pressure nozzle moved radially on a radial

track on the left end of the rotating shells could clean all the boiler shells automatically.

51. PAGE 21, LINE 21:

Applicants anticipate that the cleaning nozzles would operate during normal operation of the system. If desired, compressor **220** could be stopped occasionally to stop vaporizing incoming liquid. The flow of incoming liquid could also be stopped. Shell rotation would continue, however. The loosened sludge then flows towards the boiler-vapor exit side of the shells and is collected as described. A small hydraulic accumulator could generate squirts or pulses of water at a high pressure could accomplish this task.

52. PAGE 21, LINE 27:

Though not shown in FIG. 1, the left side of the boiling surface of the shells, such as shells **40**, may have a short annular dam at the boiler exit end. The dam would be slightly higher than the planned height of the thin film. Applicants anticipate, however, that the fluid flows of incoming contaminated liquid or the concentrated contaminated liquid could be controlled such that the desired amount of fluid could be evaporated without the need for a dam.

53. PAGE 22, LINE 22:

Constructing the shells from slats allows the shell circumference and consequently the diameter to change as the number of slats varies. Thus, the width of each slat is S_1. The height of each web is S_2. Assume that 100 slats **402** and **410** form the outermost shell and assume further that that shell has a 5 ft. (1.52 m) diameter. The circumference, therefore, is 15.7 ft. (4.8 m). Each slat would be 0.155 ft. or about 1.89 in. (4.72 cm) wide. Decreasing the circumference by that amount by forming a shell with one fewer slat would result in a circumference of 15.5 ft. (4.7 m) and a diameter of 4.95 ft. (1.51 m). Thus, the circumference of a shell with one slat removed would be 0.11 ft. (1.3 in.; 3.3 cm) less. The space between the shells would be one-half that distance or about 0.7 in. (1.8 cm). As the shells get progressively narrower, the difference in diameters and, consequently spacing, changes.

Slats of different dimensions could be provided for the shells closer to the axis of rotation to compensate for this change.

54. PAGE 23, LINE 13:

If a tapered shell is warranted, FIG. 17 shows one way to make such shells. Sheet **420** is stainless steel, aluminum or another thermally conductive material and is cut into the shape shown in the figure. The shape is exaggerated in FIG. 17 to show the concept. The sheet is wrapped around a conical mandrel with edges **422** and **424** butted together and welded. The welded sheet forms a conical or tapered shape. The degree of taper varies with angle **426**. A smaller angle yields a gentler taper than a larger angle yields.

55. PAGE 23, LINE 29:

As the sludge flows over the left edge (FIG. 2) of the shells, the momentum sprays the sludge into trough **300** to be scooped out by stationary dipper tube **302**. A ventricle pump **192** forces the liquid sludge through tubing into outlet **194**. The outlet and the ventricle pump are stationary and do not rotate with the shells. An outlet in trough **190** sprays sludge into a collector for the pump. The pump pressurizes the sludge and directs it into outlet **94**. The pumped sludge then flows into manifold **196**, outlet tube **48** and outlet **50** where it is collected.

56. PAGE 25, LINE 18:

As FIGS. 2 and 3 show, a fixed radial spacing ΔR exists between each shell. In theory, the spacing can vary, but the exemplary embodiment provides fixed spacing. The following relations compute the total area (in feet) of all boiler and condenser shells:

57. PAGE 26, LINE 15:

Substituting equations (13) and (15) into equation (11) gives the total area (in square feet) of all the boiler and condenser shells, which equals:

58. PAGE 27, LINE 4:

The weight (in pounds) of the boiler and condenser shells can be determined from their known weight density, shell thickness t and surface area A_{Total}. The weight is given by:

59. PAGE 27, LINE 11:

Energy Requirements: As discussed previously, the source of energy or heat required for the boiling and condensing process comes entirely from the vapor compressor system **220**. The system pressure is adjusted to operate near a boiling water vapor pressure that is commensurate with its ambient temperature. The proposed compressor creates a pressure ratio of about 1.05 to 1.25 on the water vapor that is able to boil at ambient temperature conditions (~70°F or 21°C) about 0.5 psi (0.035 kg/cm^2) or less. Consequently, the work done by the compressor is very low. See, e.g., Keenan, J.H. and Keyes, F.G., "Thermodynamic Properties of Steam" John Wiley & Sons, 1936, pp. 28-31.

60. PAGE 27, LINE 24:

Beginning from first principles, the following discussion presents an analysis of the compressor power requirements for the distillation process. This analysis computes the work performed by the compressor on the fluid by considering the process to be adiabatic, i.e., no heat energy flowing into or out of the compressor other than what is carried by the work done on the compressed fluid. For each "cycle" of the compressor, the relative amount of heat transferred out of the compressor and its surroundings compared to the amount of heat transferred to the fluid is small and decreases with increased compressor efficiency. Inefficiency losses of the compressor result in reduced laminar kinetic flow of the vapor and increased turbulent kinetic flow. These losses are minimized with the appropriate compressor design. Still, the inefficiency losses of the compressor can still be applied to the end result to determine the approximate overall energy requirements for this distillation process.

61. PAGE 28, LINE 22:

In this expression K is a constant that can be eliminated, and γ is the ratio of specific heats at constant pressure to constant volume. The units for specific heat are (BTU/(lb °F)). The specific heat ratio γ, is defined by:

62. PAGE 30, LINE 5:

Inefficiency losses of the vapor compressor and motor subsystem increase the actual specific energy usage. If their combined inefficiency loss is ξ, the actual specific energy is:

63. PAGE 30, LINE 13:

In this expression MW is the molecular weight (pounds/mole), which has a value of 18 for water or water vapor.

64. PAGE 31, LINE 3:

In FIG. 28, the boiler pressure is p_1 or p_B and the condenser pressure is p_2 or p_C. Several families of pressure ratios (p_2 / p_1) are shown for comparison. The displayed pressure ratios are not directly apparent, but successful operation should occur with a boiler and condenser temperature differential of 4°F (2.2°C) or less. By reading the temperature values from the water vapor boiling curve of FIG. 29, temperature values can be assigned to these pressure values.

65. PAGE 31, LINE 24:

As previously mentioned, the motor and compressor inefficiencies increase actual system specific energy. See equation (29). Figure 30 shows a sample computation of the system specific energy for a wide range of efficiency losses and for several different thermodynamic cycles having varying boiler and condenser differential temperatures ΔT.

66. PAGE 31, LINE 28:

The data used in FIG. 30 are the specific energy values corresponding to a boiler temperature of 70°F (21°C) with no inefficiency losses. Large electrical motors operate with efficiencies ranging from 93% to 95%. Therefore, 94% is a rea-

sonable motor efficiency ξ_M. Likewise, jet aircraft compressors have efficiencies of 85% to 90% or better. Thus, one expects a conservative compressor efficiency ξ_C of 85%. The combined overall system efficiency should be equal to or greater than the following:

67. PAGE 32, LINE 19:

The power costs required for the processes of the present invention can be computed by noting that the primary power source is the compressor shaft power. The system also requires an additional small amount of energy for rotational acceleration of the fluid in the boiler/condenser shells. Applicants estimate that additional energy to be approximately 0.2 W hrs/lb for the exemplary 350 gallon/day system.

68. PAGE 33, LINE 1:

One acre-foot = 1.23 x 10^6 liters. In this expression E_ρ^ξ is the system specific energy (FIG. 22) in Watt-hours per pound and $C_¢$ is the cost of electrical power that the utility charges in cents per kilowatt-hour.

69. PAGE 34, LINE 26:

The general relation that describes the boiling heat transfer coefficient in both the pool and nucleate boiling regimes is based on work by Rohsenow (*See,* for instance, *Handbook of Applied Thermal Design*, Eric C. Guyer, Editor in Chief, McGraw Hill 1989, pp. 1-79). Guyer references the original work of Rohsenow performed in 1952 using correlated experimental data. The relation developed by Rohsenow has the following general form:

70. PAGE 34, LINE 26:

Results of Boiling Heat Transfer Calculations: Figures 19-24 show a series of computations of the general trends, which occur with rotational g's on boiling heat transfer rates. These figures are both three-dimensional surface renditions and two-dimensional slice plots for two different boiler surface material conditions and three different ambient input temperatures. Some of the figures show the enhanced boiler heat transfer behavior with increased rotational g for a roughly coated Teflon®

PTFE surface on a stainless steel shell at ambient input temperatures of 70°F, 90°F, and 110°F. This series employs a material characteristic coefficient that was empirically determined to be C_{SF} = 0.0058 for a rough Teflon coat on stainless steel. *See,* Guyer, pp. 1-79. When using rough Teflon coated stainless steel instead of polished stainless steel, a 262% increase in boiling heat transfer occurs. The second series shows similar results for a different boiler surface, namely polished stainless steel only. Here, the characteristic surface coefficient is C_{SF} = 0.0080. Physically, the performance is reduced for polished stainless steel compared to a rough surface-coated Teflon on stainless steel because the roughened Teflon surface provides more locations for nucleate boiling sites. Surface wetting characteristics increase with the Teflon coating.

71. PAGE 36, LINE 7:

Using anodized aluminum with a Teflon coating on the boiler is another possibility. First, aluminum reduces the tangential hoop stress in proportion to the ratio of material densities, and the density of aluminum is about 1/3 that of stainless steel. Consequently, the tangential stress decreases from 7,600 psi at 850 g's to about 2,525 psi for corrosion resistant aluminum. Aluminum also has very high thermal conductivity compared to stainless steel. Aluminum has a thermal conductivity of about 120 (BTU/(hr ft °F)) compared to stainless steel that has a value of about 10 (BTU/(hr ft °F)). The importance of high thermal conductivity in the wall material becomes greater when the heat transfer coefficients due to boiling and condensing become less of a limiting mechanism.

72. PAGE 37, LINE 20:

To enhance the condensate heat transfer coefficients further, vapor flow should not shear with the condenser surface. Otherwise, it is possible for surface filming to build up. *See,* e.g., Singer, R.M. and Preckshot, G.W., *The Condensation of Vapor on a Rotating Horizontal Cylinder*, Proceedings of the 1963 Heat Transfer and Fluid Mechanics Institute, June 1963. To help eliminate the differing relative velocities of the vapor condenser interface, it may be feasible to insert longitudinal

plastic tube separators inside the condenser chambers. These tubes act to compartmentalize the vapor flow into channels and to impart a tangential (rotational) velocity on the condensate fluid. These compartments help impart angular momentum to the condensate so that the g forces immediately remove the condensates from the condenser surface.

73. **PAGE 38, LINE 26:**

Boiler Shell Stress and Strain - Overview: High velocity rotation induces shell stresses from the shells' own weight and from the weight of the rotating fluid against the shells. Calculations show that typical stress values are very low (7.5 ksi to 12 ksi) even up to 1,000 g's. Summary graphs of FIGS. 25 and 26 plot the rotational pressure variation exerted against the shells from the weight of the fluid for different liquid film thickness. These pressures are used to compute the total stress against the shell.

74. **PAGE 39, LINE 14:**

FIGS. 25 and 26 show the calculated stress, strain and rotational g's on the outer boiler condenser ring for a 5 ft. (1.5 m) shell diameter. The graphs show that the induced tangential stresses in the outer shell (inner shells have lower stresses) due to high rotational speeds are negligibly small compared with typical maximum acceptable stress for a 0.015 in. (0.38 mm) thick stainless steel shell. Figure 26 shows that even with the additional load of the liquid film carried by the shell (assumed to be 0.070 in. (1.8 mm) thick–much thicker than expected), the maximum tangential stresses (which occur at the shell's inner surface) are still easily within design limitations. The radial growth of the shell is also very small with a maximum growth of only about 2 mils at the outer 5 ft. diameter. *See*, FIG. 26.

75. **PAGE 46, LINE 2 — ABSTRACT OF THE DISCLOSURE:**

The present water reclamation system comprises a series of concentric thin shells. The shells mount within a housing that can be maintained under vacuum or low pressure. The shells rotate at high velocity. Contaminated liquid from outside the housing is injected into the space between half the shells. The centrifugal force

causes the liquid to form a thin film along the inward facing surface of the shell. A compressor lowers the pressure adjacent the thin film causing the liquid to boil. The compressor carries the vapor to the other side of those shells at a slightly higher temperature. There the vapor encounters the wall, which is cooler and its heat is transferred to boil the contaminated liquid. The vapor condenses, and rotation throws the condensate against the adjacent wall where it is collected. When condensing, the heat of condensation transfers to the shell for boiling the incoming contaminated liquid. The system needs no heat sources other than the energy of vapor compression to complete the cycle flow from vaporization to condensation. Various systems inject contaminated liquid into the device and collect the purified liquid and contaminated sludge.